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# EXPERIMENTAL PERFORMANCE OF A 2-15 KILOWATT BRAYTON POWER SYSTEM USING A MIXTURE OF HELIUM AND XENON

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ABSTRACT

A Brayton power system was operated successfully for a total of 2561 hours in a vacuum environment. Two rotating units were tested and compared. With the exception of the heat source and heat sink, the components were flight-type hardware. No major technological problems occurred during system operation.

With the design working fluid (a mixture of helium and xenon with a molecular weight of 83.8), an alternator gross output of 11.9 kilowatts was obtained. The corresponding gross and net conversion efficiencies were 0.33 and 0.29 respectively. Data are presented for turbine inlet temperatures from 1300° F to 1600° F, compressor outlet pressures from 25 psia to 44 psia, and compressor inlet temperatures from 45° F to 94° F.

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## SUMMARY

A Brayton power system was operated successfully for a total of 2561 hours in a vacuum environment. Two rotating units were tested and compared. With the exception of the heat source and heat sink, the components were flight-type hardware. No major technological problems occurred during system operation.

With the design working fluid (a mixture of helium and xenon with a molecular weight of 83.8), an alternator gross output of 11.9 kilowatts was obtained. The corresponding gross and net conversion efficiencies were 0.33 and 0.29 respectively. Data are presented for turbine inlet temperatures from $1300^{\circ}$ F to $1600^{\circ}$ F, compressor outlet pressures from 25 psia to 44 psia, and compressor inlet temperatures from $45^{\circ}$ F to $94^{\circ}$ F.

## INTRODUCTION

The NASA-Lewis Research Center is presently developing a closed-loop Brayton cycle system (ref. 1 & 2) for space applications. This system was designed to deliver from 2 to 10 kW of electrical power continuously and to have shutdown and restart capability. At the present time, ground tests of all major components, subsystems, and the complete power conversion system, without nuclear heat source and waste-heat radiator, have been or are being conducted.

Component tests have indicated that the system will be capable of producing 15 kW of gross power output with additional cooling capability for the alternator. An effort is in progress to provide the hardware required for additional cooling of the alternator.

The power conversion system was operated in a vacuum environment for a total of 2561 hours in the Space Power Facility at the Lewis Research Center. No major technological problems were encountered. The system, with the exception of the heat-source and heat-sink subsystems, consisted of flight-type hardware.

The rotating unit of the system includes a turbine, alternator, and compressor all mounted on a single shaft. The shaft is supported entirely by gas lubricated bearings. Other system components include a recuperator, a waste heat exchanger, coolant loop hardware, and associated system controls.

Two working fluids were used in the power conversion gas loop. One was the design fluid, a mixture of helium and xenon gases mixed to a molecular weight of 83.8. In addition, krypton (molecular weight of 83.8) was also used. Performance results are presented in reference 3 for the system operating on krypton with the rotating unit first tested. Reference 4 presents a limited amount of data and compares system performance for the two working fluids.

The purpose of this report is to present general performance data for the system operating with the mixture of helium and xenon. Two rotating units were tested and data for both units are compared. Experimental data are presented for variations in turbine inlet temperature from 1300 to 1600° F, compressor inlet temperature from 45 to 94° F, and compressor outlet pressure from 25 psia to 44 psia.

## BRAYTON POWER SYSTEM DESCRIPTION

A schematic of the power system is presented in figure 1. The components comprising the gas loop are the heat-source heat exchanger, the Bratyon rotating unit (BRU), and the Brayton heat exchanger unit (BHXU). The heat exchanger unit includes the recuperator and the waste heat exchanger. The subsystems required for operation are: heat source, gas management, electrical, and the heat rejection subsystem. A photograph of the system installed in the test facility is presented in figure 2.

### Brayton Rotating Unit (BRU)

The rotating unit includes a turbine, alternator, and compressor mounted on a single shaft. During normal operation the bearings were operated in a hydrodynamic (self-acting) mode lubricated with the working gas. During startup and shutdown, external pressurization was supplied to the bearings by the gas management subsystem to maintain bearing-to-shaft clearance. The unit was installed with the shaft vertical, turbine end up. Performance characteristics of three rotating units are described in reference 5.

The design shaft speed is 36,000 rpm. The alternator design conditions are 120 volt (line-neutral), 208 volt (line-line) at 1200 hertz. Reference 6 presents experimental results for the alternator and required controls.

## Brayton Heat Exchanger Unit (BHXU)

The heat exchanger unit includes a recuperator, a waste heat exchanger and ducting required to connect to the rotating unit. The recuperator is a gas-to-gas counterflow unit. The waste heat exchanger is a gas-to-liquid cross-counterflow unit. Plate and finned-surface construction is used for all flow passages. The waste heat exchanger has redundant coolant flow passages, though only one liquid loop was used at any time. The design and performance results of two Brayton heat exchanger units are presented in reference 7.

## Gas Management Subsystem

This subsystem supplies working fluid for gas injection starts and hydrostatic support of bearings during startups and shutdowns. The subsystem also provides for changes of gas loop pressure.

## Electrical Subsystem

The electrical subsystem regulates the alternator output voltage and shaft speed; distributes alternator power among the user's load buss, a parasitic load resistor, and a dc power supply for system requirements; and controls the system during startups, steady-state operation, and shutdowns. The performance of the electrical subsystem is reported in references 8 and 9.

## Heat Rejection Subsystem

A silicone liquid, Dow Corning 200 (2 cs at 77° F), was circulated through three parallel paths to remove heat from the waste heat exchanger, the alternator, and the electrical system packages which were mounted on coldplates. For added reliability, two identical cooling loops were installed. During normal operation one loop was active and the other was inactive. The pump and motor in each loop are constructed as a sealed assembly.

### Heat Source Subsystem

In the source heat exchanger the working fluid passed through a bank of 40 U-tubes in parallel. The tubes were radiantly heated.

## TEST FACILITY

The test was conducted in the NASA Space Power Facility located near Sandusky, Ohio. The Space Power Facility is one of several test facilities forming the Plum Brook Station of the NASA Lewis Research Center. A 100-foot diameter by 120-foot high aluminum vacuum chamber, within a concrete enclosure, is the principal feature of this facility (fig. 3). The concrete enclosure provides shielding for the conduct of nuclear experiments as well as withstands the majority of the atmospheric pressure differential when the chamber is evacuated. Experiment access to the chamber from the adjacent assembly and disassembly areas is provided by two 50 by 50-foot doors in the chamber and enclosure, respectively.

Vacuum levels in the $10^{-6}$ torr range have been achieved since the facility became operational in 1969. The Brayton power system was located in the center of the 100-foot diameter chamber for testing. All performance testing was conducted under vacuum conditions and represented the first operational test program of the facility and the Brayton power system.

## INSTRUMENTATION

The instrumentation installed for the test was divided into two categories: control and development. Control instrumentation consisted of those measurements required for the control, monitoring, and operation of the power system.

Development instrumentation consisted of sensors installed in addition to the control sensors to determine overall system performance as well as gross component performance.

The performance data presented in this report were obtained through the measurements listed in table I. All pressure measurements were made using static pressure taps with strain gage pressure transducers connected. Total pressure probes and rakes were not required due to the low velocities in the gas loop. Temperature measurements were made using either iron-constantan or Chromel-Alumel thermocouples depending on the location. Both surface and stream (probe) types were used as required. Liquid coolant flow measurements in the heat rejection system were made using turbine-type flowmeters. Gas loop flow was measured by a short, high recovery Venturi (Dall tube).

The data were recorded using the facility digital data acquisition system.

## OPERATING EXPERIENCE

The Brayton power system was operated for a total of 2561 hours. This total time was accumulated on all system components except the rotating unit (BRU). The bearings of the first BRU were damaged at 668 hours of operation due to a 44 percent shaft overspeed. The overspeed was caused by a 3-phase short circuit in the alternator power connector when low-temperature solder, used in the connector, melted. Corrective action was taken, the BRU was replaced with an identical unit that had accumulated 1000 hours of operation in a separate test, and no further problems were encountered. The initial 584 hours of operation with the first BRU were with krypton and the final 84 hours were with the helium-xenon mixture. With the second unit, BRU-2, the system was operated for 1893 hours, 358 hours with krypton and 1535 hours with helium-xenon. During this test period 1214 hours of continuous operation were achieved with helium-xenon (over 1100 hours continuous with a turbine inlet temperature of 1600$^{\circ}$ F).

During early testing of the system with the first rotating unit, problems with test support equipment limited the turbine inlet to 1500$^{\circ}$ F. The problems were corrected and the design turbine inlet temperature (1600$^{\circ}$ F) was obtained with the second rotating unit.

## PERFORMANCE RESULTS

The system performance was mapped by varying the three independent parameters: turbine inlet temperature (1600$^{\circ}$ F design value), compressor inlet temperature (80$^{\circ}$ F design value), and the compressor outlet pressure (system design range from 14 psia to 45 psia). The design operational range of compressor outlet pressure was intended to provide gross alternator outputs from 2.25 to 10.5 kilowatts at the design temperatures.

The symbols used are presented in Appendix A and the methods of calculation are presented in Appendix B.

## Varying Compressor Outlet Pressure

The effect of compressor outlet pressure on alternator gross output and system gross conversion efficiency can be determined from figure 4. At the design turbine inlet temperature of $1600^{\circ}$ F the gross output, measured at the alternator terminals, varied from 6.3 kilowatts at 25.7 psia to 11.9 kilowatts at 44.4 psia. For these conditions the gross efficiency, the ratio of gross alternator output to thermal power added, increased from 0.30 to about 0.33. At a turbine inlet temperature of $1300^{\circ}$ F the alternator gross output varied from 3.6 kilowatts at 25.8 psia to 7.5 kilowatts at 43.7 psia. The corresponding values for gross conversion efficiency are 0.20 and 0.24. The rate of change of alternator gross output with respect to compressor outlet pressure ranged from about 0.29 kW/psia at $1600^{\circ}$ F turbine inlet to about 0.21 kW/psia at $1300^{\circ}$ F turbine inlet.

All of the alternator gross output is not available to a user. Components in the electrical subsystem and the coolant pumps require electrical power to maintain system operation. The total requirement is a relatively constant housekeeping load of approximately 1150 watts plus a small parasitic load residual to accommodate slight variations in the system operating point (ref. 8). Assuming 250 watts of parasitic load residual, the net power available to a user would then be the alternator gross output minus 1.4 kilowatts. The net power and net conversion efficiency are presented in figure 5. The net efficiency is defined as the ratio of net power available to the thermal power added. At a turbine inlet temperature of $1600^{\circ}$ F the net power varied from 5.1 kilowatts at 26 psia compressor outlet pressure to 10.5 kilowatts at 44.4 psia. The corresponding values for net conversion efficiency are 0.24 and 0.29.

## Varying Turbine Inlet Temperature

The effects of varying turbine inlet temperature can be determined from figure 4. For a 100 degree decrease in turbine inlet temperature, the alternator gross power decrease ranged from about 0.9 kilowatt at 26 psia compressor outlet pressure to about 1.4 kilowatts at 44 psia. The gross conversion efficiency decreased by about 0.09 when turbine inlet temperature was reduced from 1600 to 1300° F.

### Varying Compressor Inlet Temperature

The alternator gross output and gross conversion efficiency are presented as a function of compressor inlet temperature in figure 6. Data were obtained for turbine inlet temperatures from 1300 to 1600° F. For all turbine inlet temperatures, a 10° F increase in compressor inlet temperature resulted in a decrease of about 0.4 kilowatts in alternator gross output. At 1600° F turbine inlet temperature the gross conversion efficiency decreased from 0.34 at 52° F compressor inlet temperature to 0.32 at 88° F.

The system net output and net conversion efficiency are presented in figure 7. At a turbine inlet temperature of 1600° F the net output varied from 9.3 kilowatts at 50° compressor inlet temperature to 7.5 kilowatts at 90° F. The corresponding net conversion efficiencies are 0.30 at 50° F and 0.27 at 90° F.

### Comparison of System Performance With the Two Rotating Units

A comparison of alternator gross output and gross conversion efficiency for BRUs 1 and 2 is presented in figure 8. The output and conversion efficiency for the system with BRU-2 are greater than with BRU-1 for any given turbine inlet temperature and compressor outlet pressure. At a turbine inlet temperature of 1500° F and 26 psia compressor outlet pressure the alternator gross output with BRU-2 was 0.5 kilowatt greater than with BRU-1; at 44 psia the difference was 0.9 kilowatt. The gross conversion efficiency for the system with BRU-2 was about 0.02 higher than with BRU-1.

The shaft power available to the alternator is the difference between the power developed in the turbine and the power consumed in the compressor. A comparison of gas flow rate, and temperature differences across the turbine and compressor is presented in figure 9 for the two rotating units for a 1500° F turbine inlet temperature. The gas flow rate and temperature increase across the compressor are the same for the two units. Therefore, the compressor power requirements were the same.

The gas temperature drop across the turbine for BRU-2 was about $8^{\circ}$ F larger than for BRU-1. The $8^{\circ}$ F difference represents about 0.4 kilowatt larger output for the turbine of BRU-2 at a compressor outlet pressure of 26 psia (0.77 lbm/sec gas flow rate). At a compressor outlet pressure of 44 psia (1.32 lbm/sec gas flow rate) the $8^{\circ}$ F represents about 0.7 kilowatt larger output for the turbine of BRU-2. The difference in turbine powers, as indicated by gas temperature change, was very close to the difference in measured alternator gross output. Therefore, the difference in both alternator gross output and system gross conversion efficiency was apparently due to a difference in turbine performance between the two rotating units.

## SUMMARY OF RESULTS

A Brayton cycle power conversion system was operated in a vacuum environment for a total of 2561 hours (over 1100 hours continuously at a turbine inlet temperature of $1600^{\circ}$ F). The system, with the exception of the heat source and heat sink subsystems, represented flight-type hardware. Two rotating units were tested. No major technological problems were encountered.

The system produced 11.9 kilowatts of gross power at a turbine inlet temperature of $1600^{\circ}$ F, compressor inlet temperature of $80^{\circ}$ F, and a compressor discharge pressure of 44.4 psia. For these conditions the net power was 10.5 kilowatts and the net conversion efficiency was 29 percent.

Differences in alternator power output and system conversion efficiency were apparent when data for the two rotating units were compared. The differences appreared to result from different turbine performance for the two rotating units.

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# APPENDIX A

## SYMBOLS

| | |
|---|---|
| $C_D$ | discharge coefficient for Dall tube |
| $C_p$ | constant pressure specific heat, Btu/lb mass $^{o}R$ |
| $h_w$ | pressure differential, in. of water |
| $p$ | static pressure, lb force/in.$^2$ |
| $P$ | power, kW |
| $Q$ | thermal power, kW |
| $R$ | gas constant, (ft)(lb force)/$^{o}R$ (lb mass) |
| $T$ | temperature, $^{o}R$ |
| $W_G$ | gas loop flow rate, lb mass/sec |
| $\Delta$ | difference operator |
| $\eta$ | efficiency |

Subscripts:

| | |
|---|---|
| a | alternator |
| AD | added |
| c | compressor |
| G | gas |
| GR | gross |
| n | net |
| t | turbine |
| 1 | compressor inlet condition |
| 2 | compressor outlet condition |
| 5 | recuperator cold side outlet condition |
| 6 | Dall tube |
| 10 | turbine inlet condition |
| 11 | turbine outlet condition |

## APPENDIX B

## METHODS OF CALCULATION

The following equations and definitions, along with the ideal gas relationships, were used to calculate the performance of the engine from the measured data.

The gas mass flow rate was calculated by using the following equation:

$$W_G = 1.5318(C_D)(p_5 - 0.009595\ h_w)(0.9905151 + 1.90839\ T_5\ 10^{-5})\sqrt{\frac{h_w}{P_5\ T_5}} \quad \text{(B1)}$$

where $$h_w = (P_6)(27.832) \quad \text{(B2)}$$

During the calibration of the Dall tube, the discharge coefficient was obtained as a function of the gas Reynolds number. For a first attempt to calculate the flow rate, a Reynolds number and discharge coefficient were assumed. Then with this calculated flow rate, a check of the assumed Reynolds number was made and if the two disagreed, another Reynolds number was assumed and the same process was repeated. When the difference between the assumed Reynolds number and the calculated Reynolds number was less than one percent, the flow rate value was within the accuracy of the measurements used to obtain it.

Actual turbine power was calculated as follows:

$$P_t = W_G C_{P_G}\ (T_{10} - T_{11})(1.055) \quad \text{(B3)}$$

Actual compressor power was calculated as follows:

$$P_c = W_G C_{P_G}\ (T_2 - T_1)(1.055) \quad \text{(B4)}$$

Thermal power transferred to the gas from recuperator outlet to turbine inlet was calculated as follows:

$$Q_{AD} = W_G C_{P_G}\ (T_{10} - T_5)(1.055) \quad \text{(B5)}$$

The estimated net system output power was calculated as follows:

$$P_n = P_{a_{GR}} - 1.4 \text{ kW} \tag{B6}$$

where 1.4 kW represents the estimated system internal power requirement based upon measurements obtained during system operation.

The system gross conversion efficiency was calculated as follows:

$$\eta_{GR} = \frac{P_{a_{GR}}}{Q_{AD}} \tag{B7}$$

The estimated system net conversion efficiency was calculated as follows:

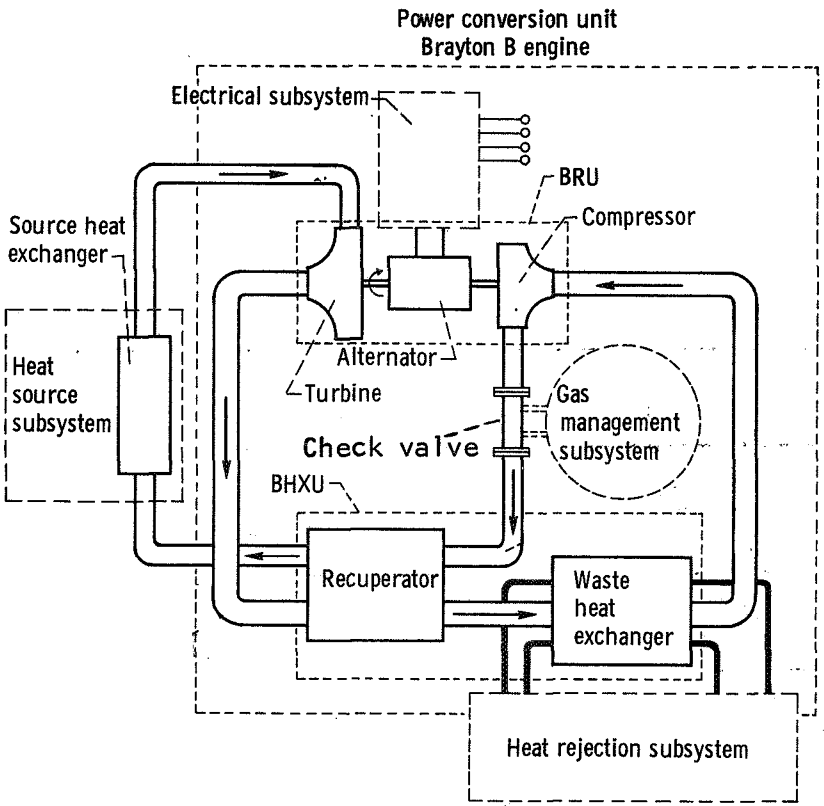
$$\eta_n = \frac{P_n}{Q_{AD}} \tag{B8}$$

TABLE I. INSTRUMENTATION USED FOR PERFORMANCE EVALUATION

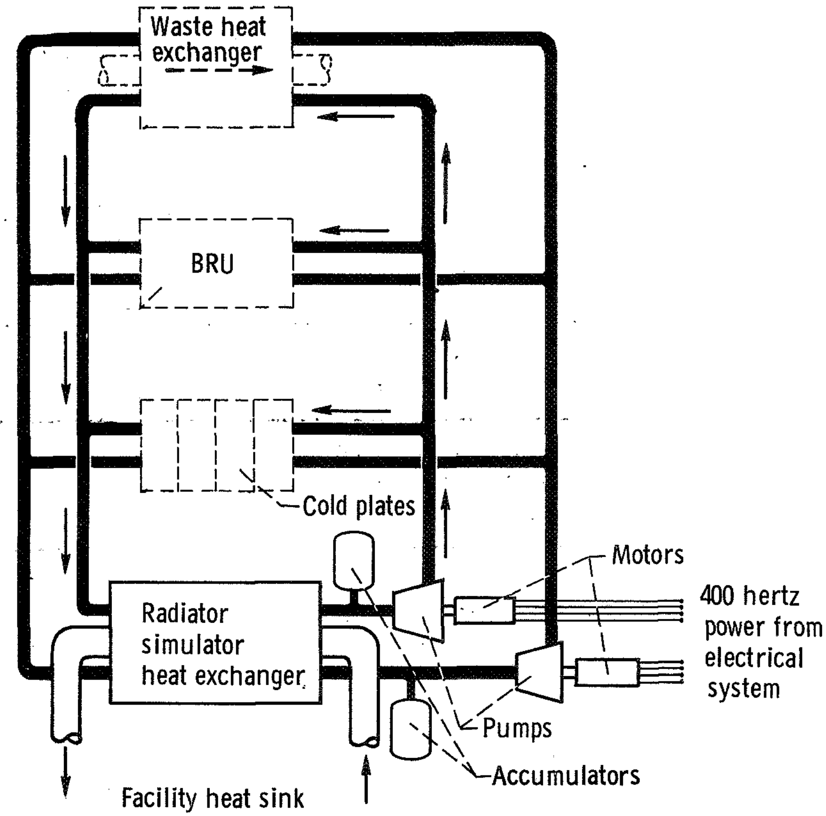
| SENSOR LOCATION & MEASUREMENT | | SENSOR QUANTITY | SENSOR TYPE |
|---|---|---|---|
| Compressor Inlet/Waste Heat Exchanger Outlet | Temperature | 2 | Probe Thermocouple (I-C) |
| | Pressure | 1 | Strain Gage Transducer |
| Compressor Outlet | Pressure | 1 | Strain Gage Transducer |
| Compressor Outlet/Recuperator Cold Side Inlet | Temperature | 3 | Probe Thermocouple (C-A) |
| Recuperator Cold Side Inlet | Pressure | 1 | Strain Gage Transducer |
| Recuperator Cold Side Outlet | Temperature | 1 | Probe Thermocouple (C-A) |
| | Pressure | 2 | Strain Gage Transducer |
| Turbine Inlet | Temperature | 3 | Probe Thermocouple (C-A) |
| | Pressure | 2 | Strain Gage Transducer |
| Turbine Outlet/Recuperator Hot Side Inlet | Temperature | 3 | Probe Thermocouple (C-A) |
| | Pressure | 1 | Strain Gage Transducer |
| Waste Heat Exchanger Inlet (Coolant) | Temperature | 3 | Surface Thermocouple (I-C) |
| | Pressure | 1 | Strain Gage Transducer |
| | Weight Flow | 1 | Turbine Type Flowmeter |
| Waste Heat Exchanger Outlet (Coolant) | Temperature | 3 | Surface Thermocouple (I-C) |

TABLE I. INSTRUMENTATION USED FOR PERFORMANCE EVALUATION (continued)

| SENSOR LOCATION & MEASUREMENT | | SENSOR QUANTITY | SENSOR TYPE |
|---|---|---|---|
| Alternator Coolant Inlet | Temperature<br>Pressure<br>Flow | 1<br>1<br>1 | Surface Thermocouple (I-C)<br>Strain Gage Transducer<br>Turbine Type Flowmeter |
| Alternator Coolant Outlet | Temperature | 1 | Surface Thermocouple (I-C) |
| Gas Flow (Dall Tube) | Pressure Differential | 2 | $\Delta$P Strain Gage Transducer |
| Alternator | Gross Power (3-Phase) | 1 | Solid State Quarter Square Multiplier |
| Vehicle Load Power | | 3 | Hall Effect Wattmeters with Output from Each Phase Summed for Total Power |
| Rotating Unit Speed | Speed | 2 | Capacitance Probe, Counting Pulses per Revolution |




Schematic diagram, Brayton power system.




Heat rejection subsystem schematic diagram.

Figure 1. Schematic diagrams of Brayton power system and Heat rejection subsystem

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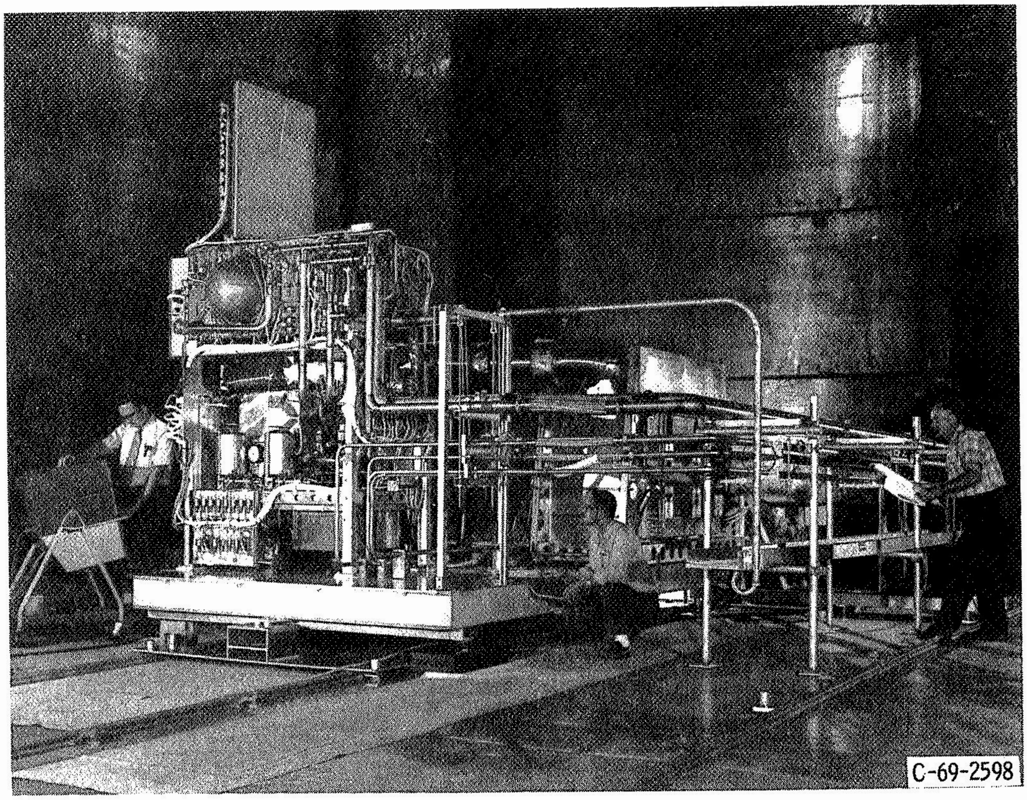


Figure 2. - Brayton system installed in SPF.

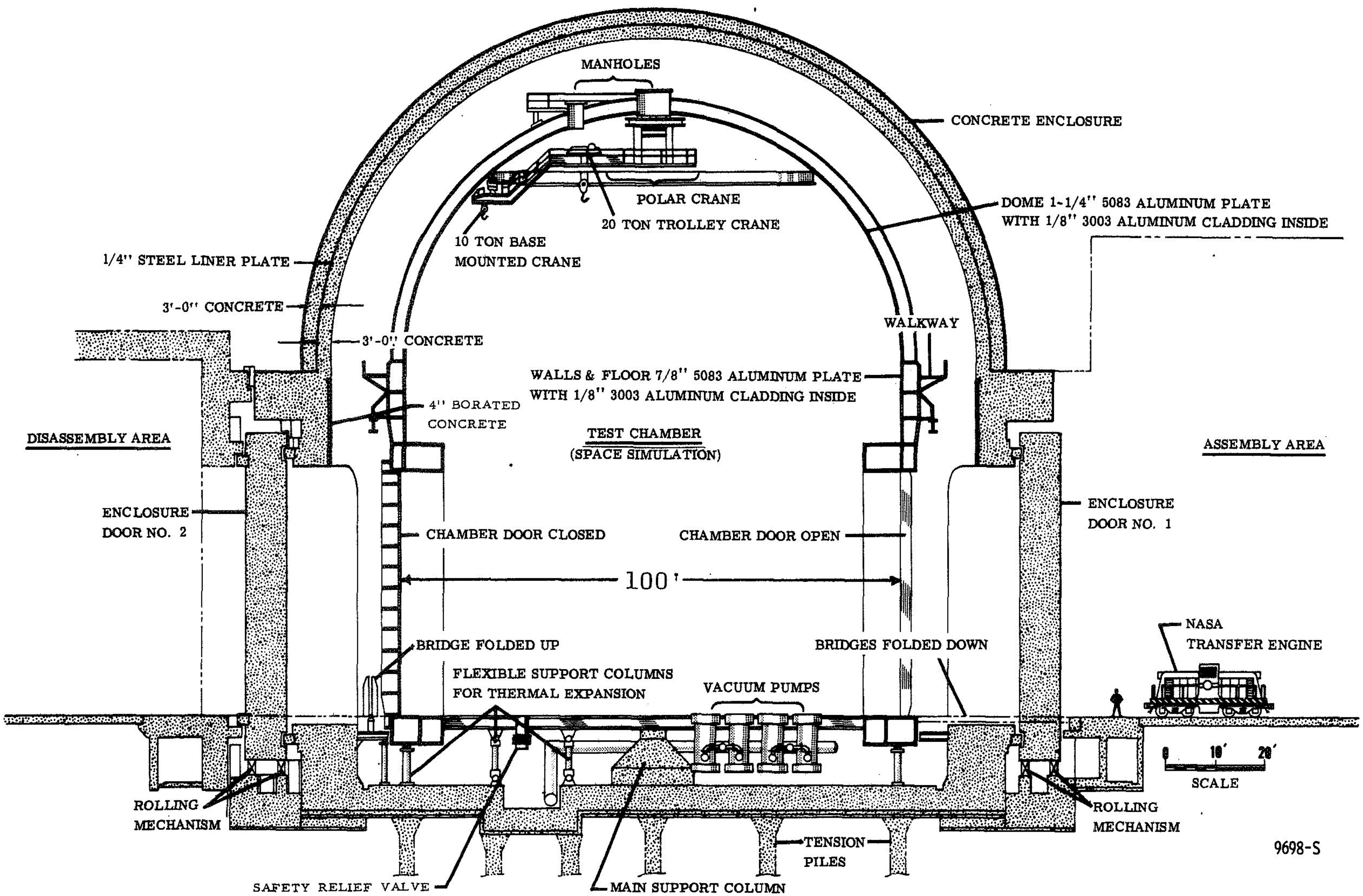


Figure 3 - Cross section through Test Chamber - Space Power Facility

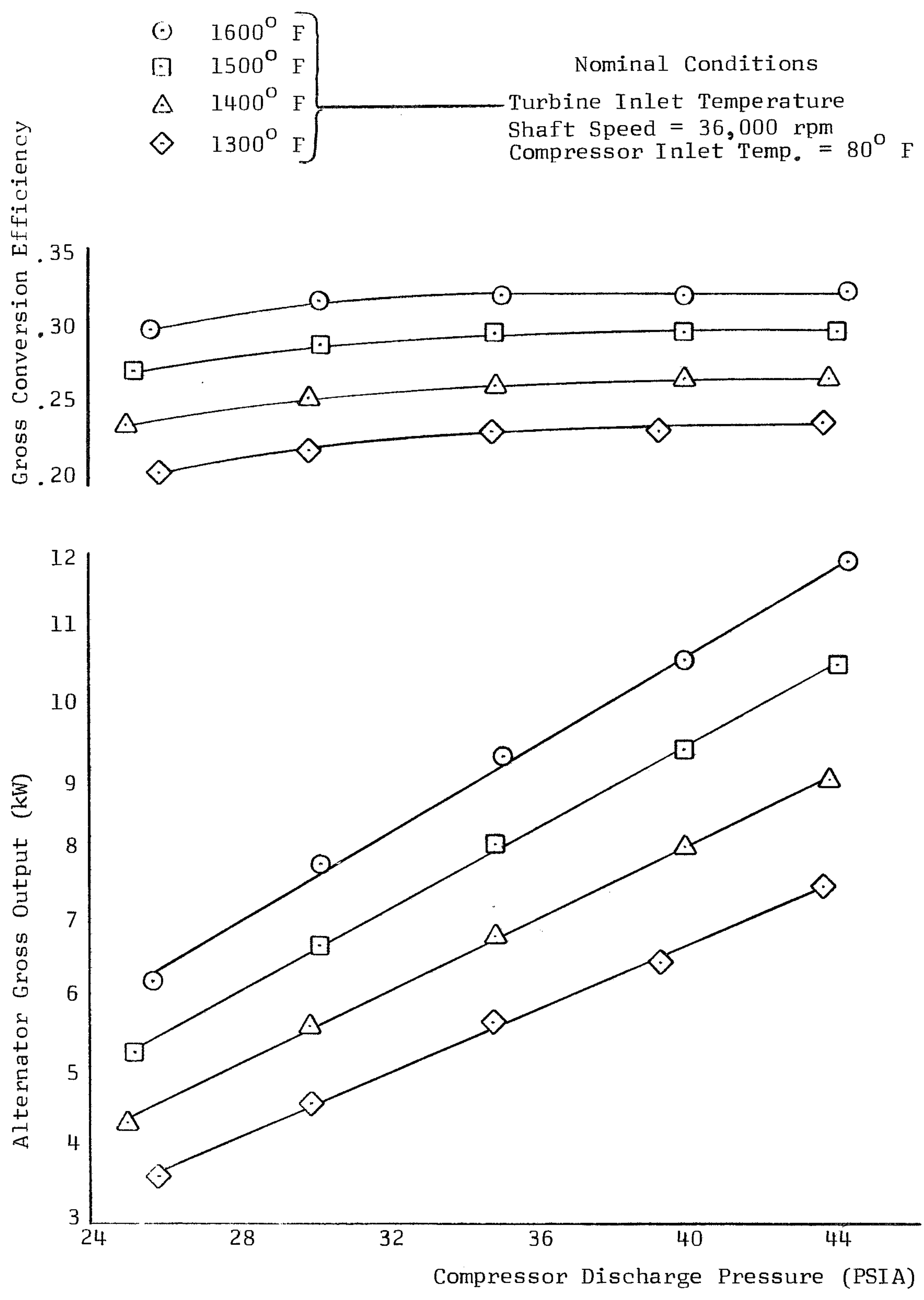


FIGURE 4. System Gross Performance with Rotating Unit BRU-2

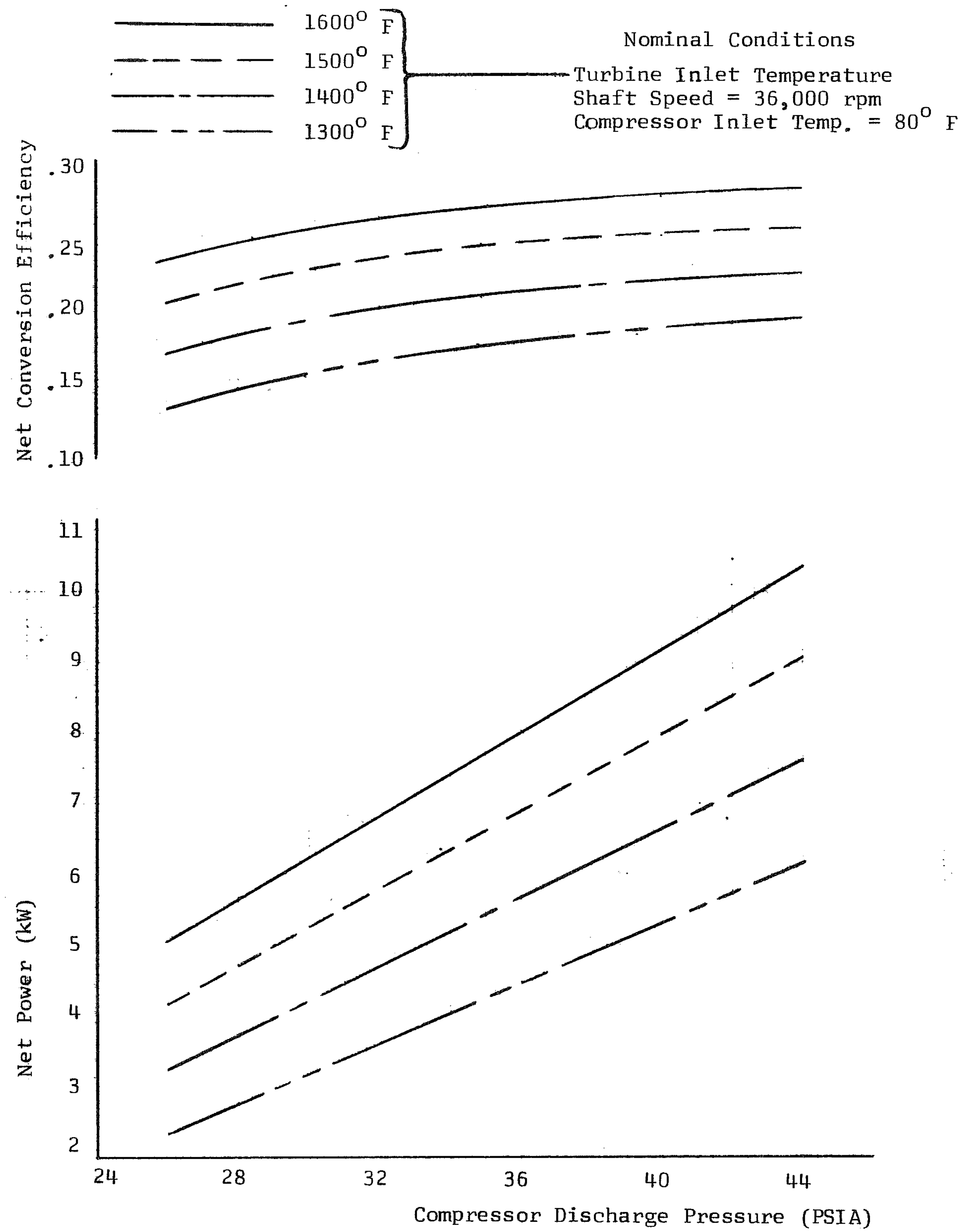


FIGURE 5. System Net Performance with Rotating Unit BRU-2

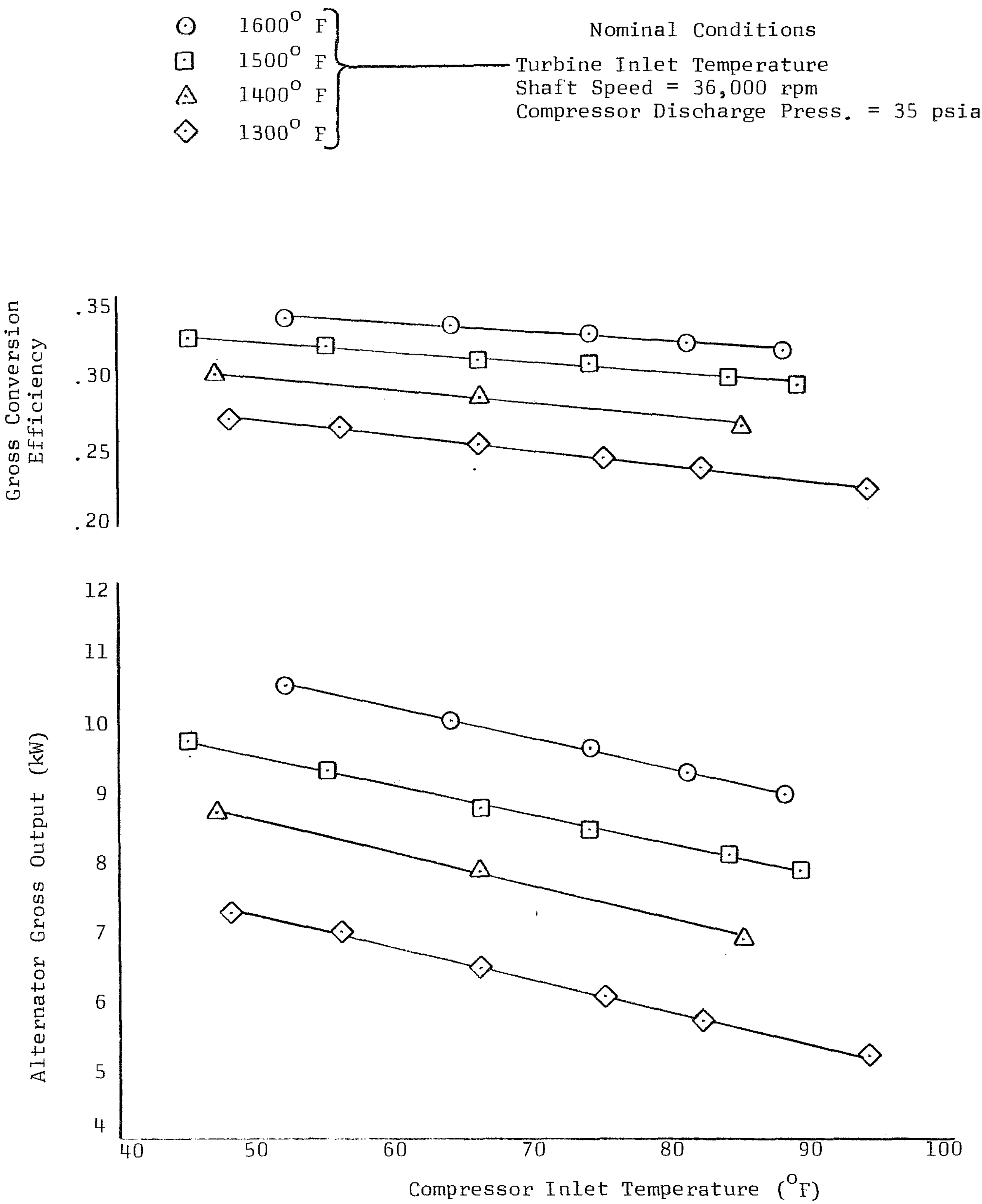


FIGURE 6. System Gross Performance with Rotating Unit BRU-2

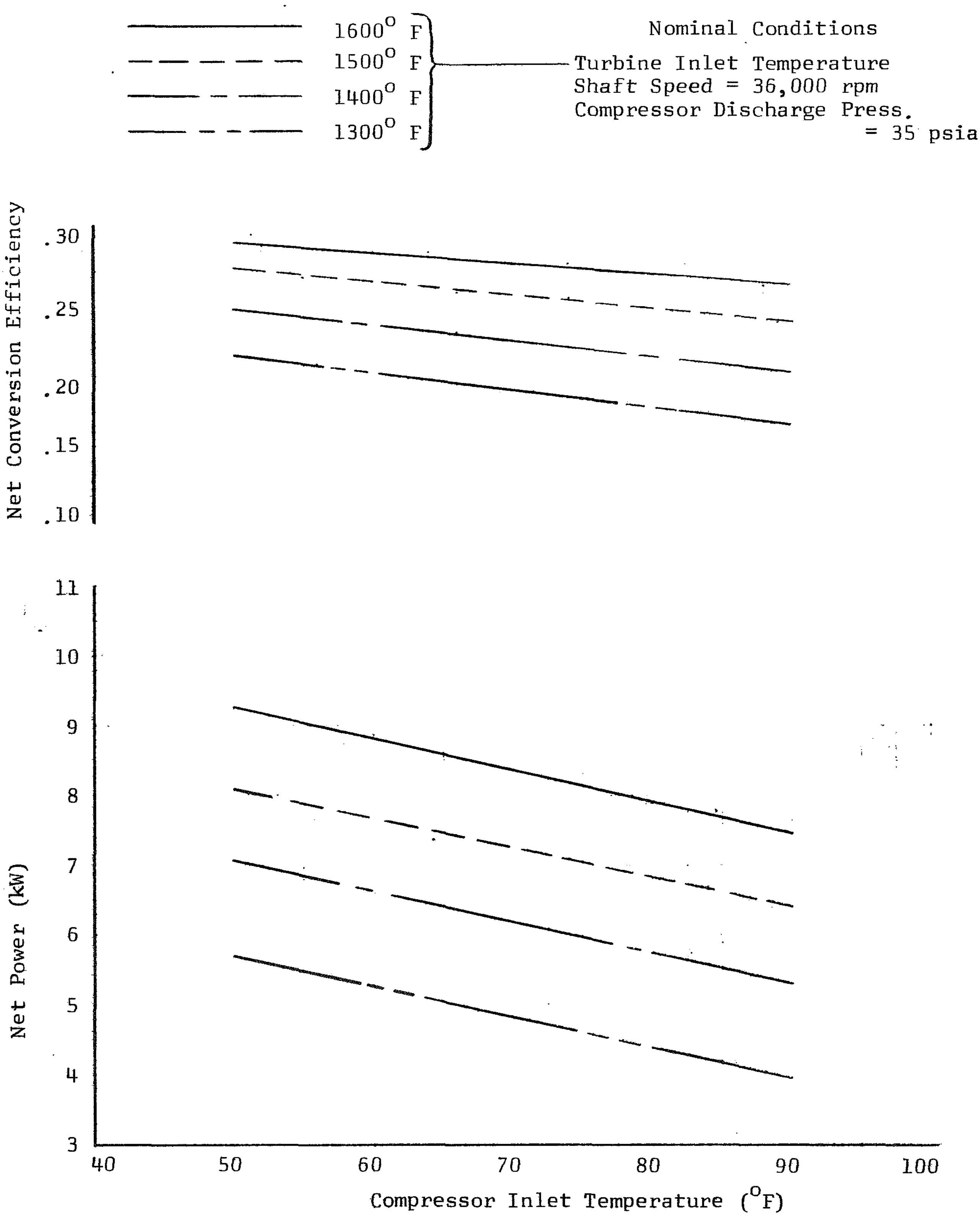


FIGURE 7. System Net Performance with Rotating Unit BRU-2

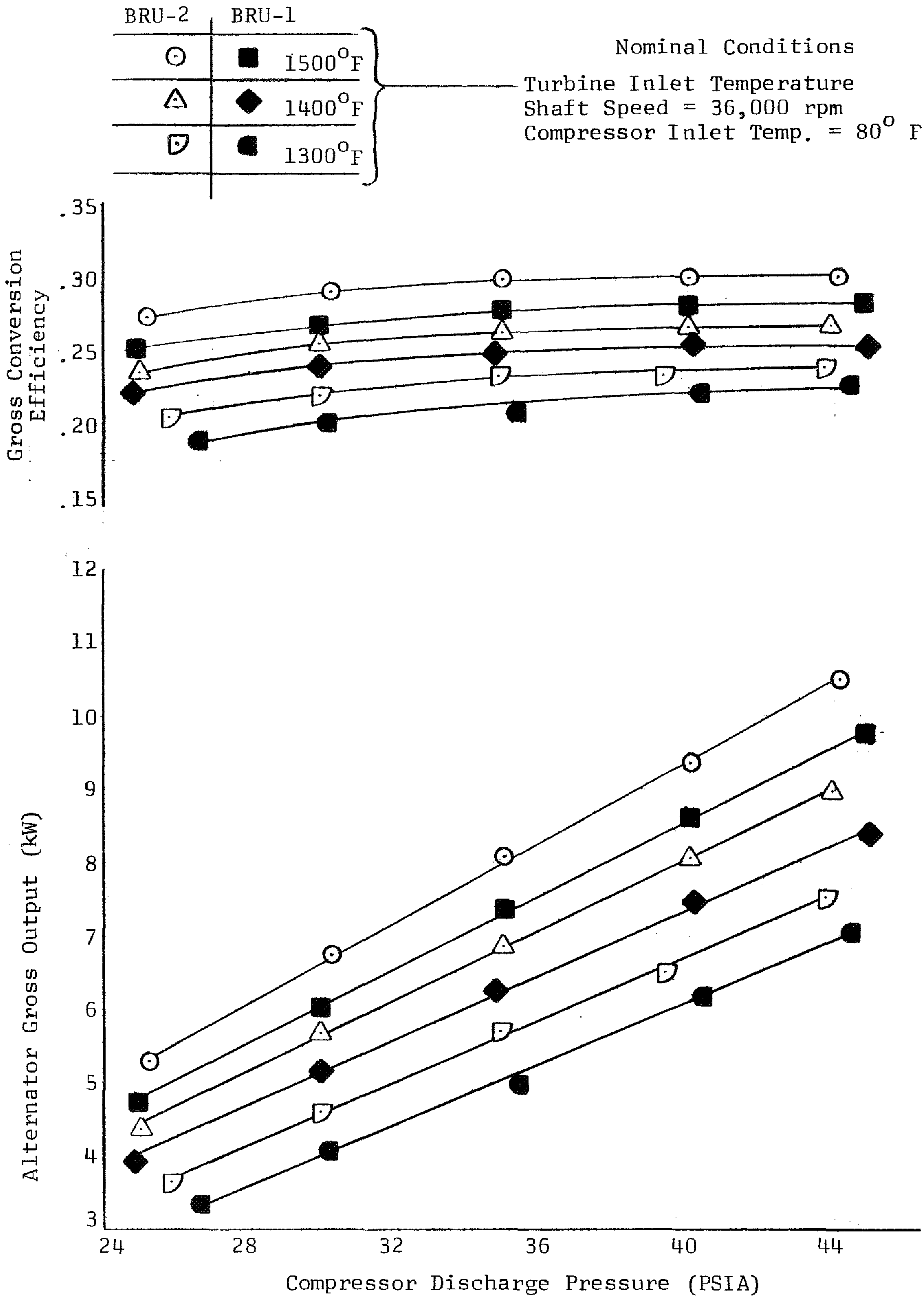


FIGURE 8. Comparison of System Performance with Rotating Units BRUs 1 and 2 for the Helium-Xenon Working Fluid

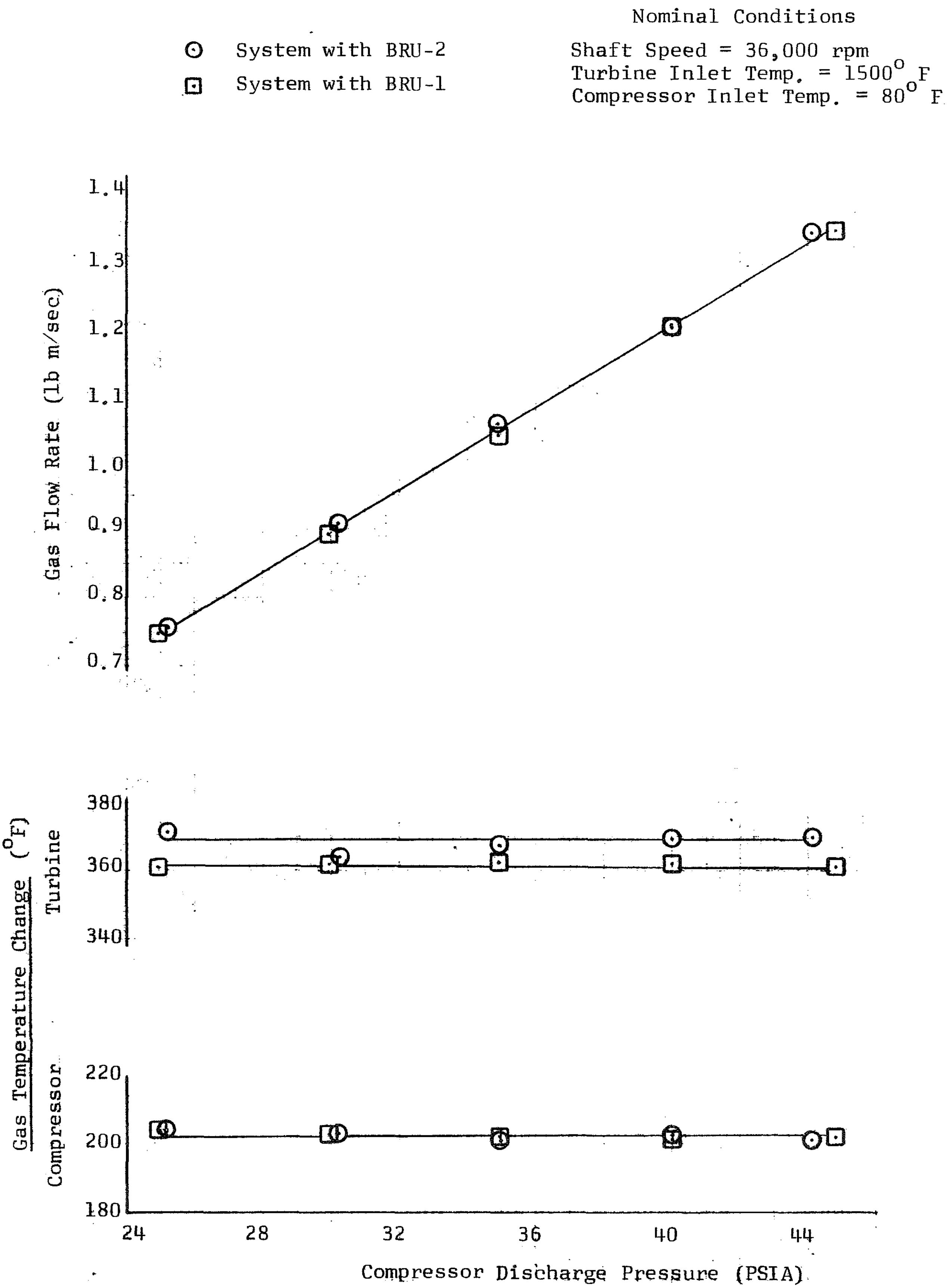


FIGURE 9. COMPARISON OF ROTATING UNITS BRUS 1 AND 2